CHAPTER 5

WASTE HEAT DISPOSAL

5-1. Heat sinks.

a. In a hardened installation, the designer will provide a means of transferring the waste heat from the equipment either to the atmosphere, via cooling towers, radiators, etc., during normal operations or to a hardened heat sink during the button-up period.

(1) The vulnerability to attack of surface cooling water sources such as a river, pond, or shallow aquifer will require a hardened facility to include cooling media storage that is available throughout the attack period.

(2) A number of research studies have been conducted to explore various methods of waste heat storage that are compatible with the overall requirements of the installation. These studies have considered the use of water, chilled water, chilled brine, rock, ice, and soil as heat sink materials. Water and water/glycol systems have been constructed and successfully tested in existing facilities.

(3) Spaces for underground water reservoirs in deep-buried facilities are usually dug in a long tunnel configuration for reasons of economy in excavation and to provide the necessary rock-surface area for heat transfer. In the calculations that follow, the tunnel shape is assumed and the cylindrical approximation is used.

b. When evaluating heat sinks, it is worthwhile to consider the use of hardened diesel fuel storage for extra heat sink capacity. The economics of installing extra equipment and controls in the fuel oil system will be balanced against increasing the size of the water-heat sink. Heating the diesel fuel will increase the efficiency of the prime mover, but there are some disadvantages.

(1) The specific heat of diesel fuel is approximately 0.5 Btu/lb ^{0}F, which is half that of water. Diesel fuel weighs 7.0 to 7.5 lb/gal, while water weighs 8.34 lb/gal. Thus, a gallon of diesel fuel has less than half the heat absorbing capacity of a gallon of water.

(2) No. 2 diesel fuel has a flash point of approximately 125 ° F. As the fuel reached this temperature, some of the more volatile components of the fuel will vaporize and create a safety hazard. These safety considerations limit the temperature of diesel fuel sinks to a maximum temperature of 900 F to 1000 F.

c. For convenience, and unless otherwise noted, the coolant fluid will be termed “water” in the remainder of this chapter. In the equations, however, the density and the specific heat of the coolant will be indicated by the usual symbols but with an asterisk to permit calculations with other coolant fluids than water.

(1) The maximum allowable coolant temperature is a major consideration in the design of heat sinks. Condensers of refrigeration equipment allow an entering water temperature as high as 100 “F to 1100 F. Higher temperature will reduce efficiency and are damaging to the refrigeration equipment. Heat exchangers of prime movers such as diesel engines will allow entering water temperatures up to 160 ^{0}F. or higher.

(2) Because of the difference in the allowable water temperature rises, separate reservoir systems will be designed to receive heat from the two separate heat sources. The two reservoirs will be proportioned in their respective loads, so that when the diesel reservoir is emptied, the engine can be cooled by water wasted from the AC compressor unit condensers, until this supply has also been exhausted.

5-2. Once through and recirculated reservoirs.

a. Waste heat from the power or weapon system equipment may be rejected by drawing water from the hardened heat sink reservoir, circulating it through the heat source one time, and then discharging the water to the outside. Normally, the water is stored at the ambient temperature of the surrounding rock; but stored chilled water may be kept in an insulated container with a small refrigeration unit to compensate for heat leak.

b. A full reservoir of volume V at temperature T_0 with allowable wasted water discharge temperature T_1 represents a total cooling capacity q_0 which, when spent at constant heat rejection rate q_0 until empty, will allow to hours of operation. Any one of these parameters may be computed from the following heat balances when the others are known:

$$Q_o = t_o q_o = V\rho^* c^* (T_1 - T_0) \qquad \text{(eq 5-1).}$$

c. If the water is recirculated from the reservoir to the engine jacket or condenser and back to the reservoir, the heat-absorbing capacity of the water is increased by the heat-absorbing capacity of the surrounding rock because the warmer water returning from the equipment increases the reservoir's temperature above that of the rock.

(1) To obtain the best effect from a reservoir used as a short-time heat sink, water will be taken from the lowest point in the reservoir and heated water discharged to the highest point. The warmest water will stratify in the upper levels, while the water taken from the bottom will be the coldest available for cooling purposes.

(2) When the water temperature reaches its maximum allowable temperature, the water of the reservoir is wasted outside the installation. The time to waste the water can be determined using equation 5-1.

(3) During the initial period when the effective capacity is enhanced by heat transfer to the rock, empirical equation 5-2 is used for the reservoir of length L, allowed to heat from T_2 to T_3 in a specified time t_1. The reservoir's constant effective heat absorption rate q_1 in Btuh during this period is then

$$q_1 = [kL/f(F_1, X)](T_3 - T_2) \qquad \text{(eq 5-2)}$$

where $f(F_1, X)$ acts as a thermal resistance factor. As in Chapter 3, this function of dimensionless parameters F_1 and X conceals the terms of the exact transient analysis, and is computed as follows

$$f(F_1, X) = .001 + (0.1)\log[1 + (F_1/F_0)^b] \qquad \text{(eq 5-3)}$$

$$F_0 = .055 + .225/X - .025\exp(-7/X) \qquad \text{(eq 5-4)}$$

$$b = 1.11 - .352\exp(-7.4/X) \qquad \text{(eq 5-5)}$$

$$X = 2\pi r_1^2 (1/S^*)(\rho c/\rho^* c^*) \qquad \text{(eq 5-6)}$$

$$r_1 = P/2\pi \qquad \text{(eq 5-7)}$$

In these equations, $F_1 - at_1/r_1^2$ is by definition the Fourier number based on t_1 in hours, X a dimensionless parameter proportional to the ratio of the heat capacitance of the rock and the reservoir, r_1 the equivalent radius in ft based on sidewall area PL of the reservoir, and the S* the reservoir coolant cross section in ft^2.

d. It may be necessary to maintain a reservoir at a temperature below the initial rock temperature to provide additional heat-absorbing capacity for an emergency period.

(1) The rate of heat removal q_2 from the water necessary to first lower the reservoir's temperature from T_4 to T_5 in a given period of time t_3 is computed from equation 5-2 replacing the temperature rise $(T_3 - T_2)$ by the temperature drop $(T_4 - T_5)$ and F_1 by $F_3 = at_3/r_1^2$ based on t_3 as shown by equation 5-8.

$$q_2 = [kL/f(F_3, X)](T_4 - T_5) \qquad \text{(eq 5-8)}$$

(2) When the desired water temperature T_5 has been reached, further heat must be continually removed from the water to offset the heat gain from the surrounding rock while maintaining the reservoir at T_5. This heat gain q_3 is proportional to the design cool down temperature drop $(T_4 - T_5)$ and decreases with time as shown by equation 5-9.

$$q_3 = kL(400/F_4)^{.31}(T_4 - T_5) \qquad \text{(eq 5-9)}$$

where $F_4 = at_4/r_1^2$ is the familiar Fourier number based on the time t_4 elapsed since the beginning of the holding period at the design temperature T_5 of the cold reservoir.

5-3. Iced Reservoirs.

a. As the term implies, iced reservoirs are reservoirs cooled down to the point where ice can coexist with the water. The heat sink capacity of an iced reservoir is greatly increased by the ice accumulated in it.

(1) As mentioned previously, the difference in acceptable cooling water temperatures for refrigeration compressors and diesel engines practically demands two reservoir systems, either or both of which may be iced.

(2) At the start of use as heat sinks, either or both of the iced reservoirs can furnish chilled water directly to unit air-conditioner cooling coils until the reservoir temperature rises about 500 F. During that time the refrigeration equipment does not need to be operated and the heat rejected to the reservoir is thus reduced accordingly.

(3) The iced reservoir should not be used as a source of chilled water for the air-conditioning system during normal operating (non-emergency) conditions, because the ice-making equipment producing ice for a reservoir operates at lower temperature and efficiencies than conventional equipment for air-conditioning,

b. If the reservoir is filled, or partially filled, with a mixture of water and ice, the water temperature will remain at or near 320 F during the addition of heat until all the ice is melted. During this period of time, and neglecting the heat gains from the rock, the mass W_1 in lb of ice, having 144 Btu./lb latent heat of fusion, represents a total heat sink capacity Q_i in Btu as follows

$$Q_i = 144\,w_1 \qquad \text{(eq 5-10)}$$

The heat gains from the rock may be neglected if the iced reservoir has been maintained at 32 "F for sufficiently long periods of time so that the heat transfer from the rock (equation 5-9) tends to zero.

c. After the ice has melted, heat will be transferred to the rock, due to the temperature rise of the water. The remaining heat sink capacity of the reservoir will be determined using equation 5-2, based on the water cross section when all the ice has melted. A reservoir filled with ice at one end only maintains an average water temperature of approximately 34 "F in the remaining length of reservoir and serves to provide an additional heat sink capacity due to sensible cooling of the water and surrounding rock below the initial temperature of the rock.

d. Ice introduced at one end of a horizontal reservoir floats, packs and jams but does not distribute itself along the length of the reservoir to a sufficient depth; therefore, some mechanical means must be made available for this purpose.

(1) The most satisfactory ice distribution method appears to be a helical-screw conveyor. This horizontal screw conveyor runs the full length of the reservoir above the maximum flotation level of the ice or a distance above the water level somewhat more than one-tenth of the depth of the water. As a result the top of the ice accumulating in any section of the reservoir will only reach the screw conveyor when underlying ice is no longer floating but is resting on the floor of the reservoir.

(2) Ice is dumped from ice making machines at one end of the reservoir. When the ice fed at the dumping point reaches the level of the screw conveyor, it is conveyed and dumped by the helical-screw into the next section, filling it to the bottom. Thus, the process repeats and the full ice front progresses in the direction of the far end.

(3) A simple pendant-level control at the far end of the reservoir that is moved by the ice front will automatically stop operation of the conveyor and ice-making equipment when the reservoir becomes completely filled.

(4) Melting of the ice front by heat transfer from the surrounding rock will allow the pendant-level control to fall to an operating position, causing more ice to be added to the reservoir. For inspection and maintenance of the screw conveyor, a walkway will be placed near the conveyor.

(5) Laboratory experiments have indicated that the best shape of ice for movement in a reservoir is cubical, spherical, or cylindrical, in pieces approximately one inch in size. Ice in crushed or flake form tends to cluster in compacted slushy masses that resist movement. Experiments have shown that the average ice volume in a water-and-ice mixture shaped in a hollow cylinder is from 40 to 50 percent, and it is probable that for non-hollow , small shapes, the ice volume percentage is materially greater.

5-4. Solid ice heat sinks.

a. When a concept is developed for an ice heat sink configuration, many relevant factors will be considered. The heat sink must be available for use when the button-up signal is given. Economy of space is a major factor in reducing the cost of deep underground excavations in rock.

(1) A block ice heat sink is available for immediate use if continuous water flow paths exist through voids between the ice blocks. However, the voids reduce volumetric utilization, Btu's stored per cubic foot of excavation.

(2) A solid ice cylinder heat sink provides for the maximum utilization of space; however, means must be provided to create an annulus of water between the ice cylinder and the sink wall. The solid cylinder must also be restrained and maintained concentric with the sink walls for proper water flow.

(3) Both types of sinks will have a spray header designed to evenly distribute incoming cooling water over the upper ice surface for uniformity of ice melting.

b. To illustrate the space economics of water, water and ice, and solid ice underground heat sinks, a comparison of Btu's that can be absorbed is made on a 100,000-gallon reservoir containing, respectively, water at approximately 8.34 lb/gal and stored at rock ambient temperature of 60 "F, a 50 percent ice/50 percent water sink, and a solid ice cylinder sink maintained at 32 F; all absorbing heat to 1600 F final temperatures, assuming ice at 7.61 lb/gal, 144 Btu/lb latent heat of fusion, and neglecting rock heat transfer.

(1) For water at 60 °F, the total capacity Q_1 is in 10^6 Btu (MBtu)

$$Q_1 = (100{,}000)\ (8.34)\ (100) = 83.4\ \text{MBtu}$$

(2) For solid ice at 32 ° F and 128 ° F rise, the total capacity Q_2 is

$$Q_2 = (100{,}000)\ (7.51)\ (128 + 144) = 204\ \text{MBtu}$$

(3) For the 50/50 sink at 32 °F, the total capacity Q_3 is

$$Q_3 = (50{,}000)\ [(7.51)\ (272) = (8.34)\ (128)] = 155\ \text{MBtu}$$

(4) Although the above comparison shows that the heat storage volumetric efficiency of an ambient water sink is approximately one-half that of a 50/50 ice and water sink and approximately two-fifths that of a solid ice sink, other factors such as space and cost for refrigeration equipment, power cost for maintaining the low temperature, and time to re-establish ice sink to design conditions after an engagement will be evaluated when the heat sink configuration is selected.

5-5. Sample problems.

This paragraph illustrated by problems the use of equations for heat sinks and discusses the relative merits of the heat-absorbing capacity of the surrounding rock as a function of time and equivalent radius.

a. Problem 1.

(1) Assume the heat-rejection rate for an underground installation during an emergency period is 2 MBtuh and that the installation must be self-sustaining for a period of 10 days. The reservoir temperature is initially 520 F and the highest water temperature permissible is 1000 F. Determine the necessary lengths of reservoirs for the cross sections given in table 5-1 with water at 62.42 lb/ft^3 density, 1 Btu/lb F specific heat and rock having 1.45 Btuh/ft ^{0}F conductivityy, 185 lb/ft^3 density, 0.2 Btu/lb ° F specific heat, and .0392 ft^2/h diffusivity.

(2) For case 1, H = W = 15 ft, and from equation 5-3 through 5-7

$$r_1 = (H + W) / 3.14 = 9.54 \text{ ft}$$

$$t_1 = 10 (24) = 240 \text{ h}$$

$$S_w = 15 (15) = 225 \text{ ft}^2$$

$$X = 2 (3.14) (9.54^2 /225) (185/62.42) (0.2/1) = 1.507$$

$$b = 1.11 - .352\exp(-7.4/1.507) = 1.107$$

$$F_0 = .055 + (.225/1.507) - .025\exp(-7/1.507) = .2041$$

$$F_1 = at_1 /r_1^2 = .0392 (240/9.54^2) = .1034$$

$$f(F_1, X) = .001 + (0.1) \log [1 + (.1034/ .2041)^{1.107}] = .0178$$

Solving equation 5-2 for L_1 with a rise $T_3 - T_2 = 48$ F

$$L_1 = 2 (10^6 /1.45) (.0178/48) = 511 \text{ ft}$$

(3) The water heat pickup Q* is the product of temperature rise, specific heat, and water mass in the reservoir. It is deducted from the total heat rejected Q_t to find the heat pickup of the rock Q_r. For case 1, t = 240h, S* = 15^2 ft^2, L_1 = 511 ft, and with coolant heat capacity p*c* = 62.42 Btu/ft^3 F and ($T_3 - T_2$) = 48 F rise.

$$Q_t = qt = (240) (2) = 480 \text{ MBtu}$$

$$Q^* = 62.42 (LS^*) (T_3 - T_2) = (62.42) (511) (15^2) (48) = 344 \text{ MBtu}$$

$$Q_r = Q_t - Q^* = 480 - 344 = 136 \text{ MBtu}$$

$$Q_r /Q_t = 1 - (Q^*/qt) = .282 \text{ or } 28\%$$

(4) The results of similar computation for cases 2 and case 3 are shown in table 5-1 with reservoir lengths computed as follows:

$$L_2 = 2(10^6 /1.45) (.01065/48) = 306 \text{ ft}$$

$$L_3 = 2(10^6 /1.45) (.00733/48) = 211 \text{ ft}$$

(5) Table 5-1 shows that for a reservoir of smaller cross-section and longer length, a greater fraction of the heat is absorbed in the surrounding rock, and this fraction is approximately proportional to

the surface area in contact with the water. Also, the volume of water or amount of excavation is less for the reservoir with the smallest cross-section.

(6) If a reservoir is to be used ultimately as a heat sink for the waste heat from a prime mover such as a diesel engine, the maximum water temperature can probably be higher than 1000 F. For case 2 with other conditions the same but with the maximum allowable water temperature equal to 1600 F instead of 1000 F, the length of the reservoir adjusted proportionally to the temperature differential would be

$$L = 306\,(48/108) = 136\text{ ft}$$

The volume of reservoir reduces then to 54,400 ft^3, but the percentage of the total heat input absorbed by the rock does not vary. The reason is that this percentage is governed by the time available for heat absorption by the rock for a reservoir of a given equivalent radius.

TABLE 5-1

Comparison of 10 Day Capacity Heat Sinks

Parameter, Symbol	Unit	Case 1	Case 2	Case 3
Width of Reservoir, W	ft	15	20	20
Height of Reservoir, H	ft	15	20	30
Equivalent Radius, r	ft	9.54	12.73	15.91
Operating Time, t	h	240	240	240
Cross-section, S^*	ft^2	225	400	600
Equation 5-6, X	--	1.507	1.509	1.572
Equation 5-5, b	--	1.107	1.107	1.107
Equation 5-4, F_0	--	.2041	.2038	.1978
Fourier Number, F_1	--	.1034	.0581	.0371
Resistance Factor, $f(F_1,X)$	--	.0178	.0106	.0073
Length of Reservoir, L	ft	511	306	211
Total Heat Rejected, Q_t	MMBtu	480	480	480
Water Heat Pickup, Q_w	MMBtu	344	367	379
Rock Heat Pickup, Q_r	MMBtu	136	113	101
Ratio Q_r/Q_t	--	.282	.236	.210
Sidewalls, A	ft^2	30,660	24,480	21,100
Water Volume, V	ft^3	115,000	122,000	126,000

b. Problem 2.

(1) To illustrate the influence of the available time, recompute case 2 of problem 1 for 6, 8, 10, 12, and 14 days of operation. The result of the computations is shown in table 5-2 for X = 1.509, $S^* = 400\ ft^2$, $r_1 = 12.73$ ft, $q_1 = 2{,}000{,}000$ Btuh, K = 1.45 Btuh/ft °F, a = .0392 ft^2 h, and 48 °F temperature rise, or from equation 5-3 and 5-4

$$F_0 = .055 + (.225/1.509) - .025\exp(-7/1.509) = .2038$$

$$F_1 = at/r_1^2 = .0392(t/12.73^2) = t/4134$$

$$f(F_1, X) = .001 + (0.1)\log[1 + F_1/F_0)^{1.107}]$$

$$= .001 + (0.1)\log[1 + (t/842)]^{1.107}]$$

From equation 5-2 solved for L

$$L = (q_1/48\ k)[f(F_1, X)]$$

$$= (2/48)(10^6/1.45)(.001 = (0.1)\log[1 + (t/842)^{1.107}])$$

$$= 28.7 + 2874\log[1 + (t/842)^{1.107}]$$

From $Q^* = 62.42(LS^*)(T_3 - T_2)$, $Q_t = q_1 t$, and $Q_r = Q_t - Q^*$

$$Q_r/Q_t = 1 - (48/10^6)(62.42/2)(400\ L/t) = 1 - .6(L/t)$$

(2) It has been noted that the percentages of total heat absorbed by rock in table 5-2 increase as the duration of heat input increases. This shows the influence of time when the reservoir size is adjusted to yield the same temperature rise for different durations of heat input.

TABLE 5-2

Reservoirs length versus time at fixed 400 ft^2 section

Symbol	Units	Case 1	Case 2	Case 3	Case 4	Case 5
	days	5	8	10	12	14
t	hours	144	192	240	288	336
F_1	-	.0348	.0464	.0581	.0697	.0813
$f(F_1,X)$	-	.0067	.0087	.0106	.0126	.0144
L	ft	194	251	306	361	414
V	ft^3	77,600	100,400	122,400	144,400	165,600
Q_t	MMBtu	288.0	384.0	480.0	576.0	672.0
Q_w	MMBtu	232.4	300.7	366.6	432.5	496.0
Q_r	MMBtu	55.6	83.3	113.4	143.5	176.0
Q_r/Q_t		.193	.217	.236	.249	.262

c. *Problem 3.*

(1) To illustrate the advantages of reducing the heat rejection rate to the minimum, use the 194-foot reservoir of problem 2 at half the heat rejection rate or 1 MBtuh. Determine for the same temperature rise of 48 'F the operating time, the total heat capacity, and the fraction of heat absorbed by the rock.

(2) Solving equation 5-2 for time gives with $s = (T_3 - T_2)(q_1)^{-1}$

$$t = (r^2/a)(F_0)(-1 + \exp[10kLs - .01)\ln 10])^{1/b} \qquad \text{(eq 5-11)}$$

(3) For this problem $s = 48/10^6$ °F/Btuh, $F_0 = .2038$, $L = 194$ ft, $k = 1.45$ Btuh/ft °F, $r = 12.73$ ft, $b = 1.107$, $a = .0392$ ft^2/h, and the operating time based on equation 5-11 is

$$t = (842.5)(-1 + \exp[(.135 - .01)2.3])^{.903} = 313 \text{ h}$$

(4) As a result, the total heat rejection is 313 MBtu, and the heat absorbed by the water is 232.4 MBtu. This gives a balance for the rock heat pickup of 80.6 MBtu or 25.8 percent of the total. Hence, reducing the heat rejection rate increased the total heat capacity by 8.6 percent, the rock heat pickup by 45 percent, and the available time by 116 percent.

d. Problem 4.

(1) Using the parameters of problem 1 and case 2, compute the refrigeration capacity required to cool down this reservoir from 52 °F to 40 ° F in 25 days, and determine the refrigeration necessary after 60, 240, 365, and 1,095 days of holding the water at 40 °F.

(2) For the cool down, $X = 1{,}509$, $F_0 = .2038$, $a = .0392$ ft^2/h, and $b = 1.107$ remain unchanged. But F_3 must be computed for 25 days or 600 h instead of the 10 days of problem 1, so that based on $r_1 = 12.73$ and $L = 306$ ft from Table 5-1,

$$F_3 = .0392\,(600/12.73^2) = .145$$

From equations 5-3 and F_3 substituted to F,

$$f(F_3, X) = .001 + (0.1)\log[1 + (.145/.2038)^{1.107}] = .0237$$

From equation 5-8 and a drop of 52-40 = 12 °F

$$q_2 = (1.45/.0237)(306)(12) = 224{,}660 \text{ Btuh}$$

At 12,000 Btuh per ton of refrigeration, this represents a demand of 18.72 tons.

(3) For the successive holding times with the reservoir kept at 40 °F, equation 5-9 is applied. For instance, after 60 days or 1,440 h

$$F4 = .0392\,(1440/12.73^2) = .348$$

$$q_3 = 1.45\,(306)\,(400/.348)^{.31}\,(12) = 47{,}317 \text{ Btuh or } 3.94 \text{ tons}$$

(4) For holding times t_i in days, different from 60 days, the corresponding Fourier number F_i is by definition or proportional to time and, compared to F_4 above, $F_i = F_4\,(t_i/60)$. Compared to q_3 above the holding load q_i is then based on equation 5-9, approximately proportional to the reciprocal of the cube root of t_i as shown below

$$\begin{aligned} q_i &= q_3\,(F_4/F_i)^{.31} \\ &= (3.94)\,(60/t_i)^{.31} \\ &= 14/t_i^{.31} \text{ tons} \\ &= 2.55 \text{ tons for } t = 240 \text{ days (4 months)} \\ &= 2.25 \text{ tons for } t = 365 \text{ days (one year)} \\ &= 1.60 \text{ tons for } t = 1{,}095 \text{ days (3 years)} \end{aligned}$$

e. Problem 5.

(1) This problem illustrates the case of the reservoir of problem 4, but in this case it contains 40 percent ice by volume. Below 50 °F water from the reservoir is used as chilled water in the system. From 50 °F to 100 °F this water is used as condenser water for the refrigeration equipment. The heat rejected to the sink is 2 MBtuh with the refrigeration equipment operating, 20 percent of which corresponds to the energy input required to drive this equipment. The heat rejection is 1.6 MBtuh as long as the sink temperature does not exceed the 50 °F temperature level. Determine how long the iced reservoir can be utilized under these conditions.

(2) For an ice density of 57.5 lb/ft^3 the ice melting latent heat capacity is

$$Q_i = 20\,(20)\,(306)\,(0.4)\,(57.5)\,(144) = 405.4 \text{ MBtuh}$$

At 1.6 mmBtuh heat input rate the time t_i it takes to melt the ice is

$$ti = 405.4/1.6 = 250 \text{ h}$$

(3) The ice density is only 92 percent that of water. After all the ice is melted the volume filled by water in the reservoir will increase by 0.92 (.40) or 36.9 percent or a total of 96.9 percent. The water cross-section is then 400(.969) = 387.5ft^2 and X adjusted for less than full is

$$X = 1.507\,(400/387.5) = 1.556$$

(4) Then F_0 = .1993 from equation 5-4 and b = 1.107 (equation 5-5) for 32 °F to 50 °F or 18 °F rise, q = 1.6 MBtuh, k = 1.45 Btuh/ft °F, L = 306 ft, r and a are unchanged, $s = (T_3 - T_2)q^{-1} = 11.25/10^6$, and the operating time calculated using equation 5-11 with (10kLs - 0.01) = .0399 and $F_o r^2/a$ = 823.9 is

$$t_1 = 823.9\,[-1 + \exp(.0399 \ln 10)\,]^{.903} = 99 \text{ h}$$

(5) From 50 °F to 100 °F the rise is 50 °F, q = 2 MBtuh, and s = $(50/2)\,10^{-6}$. With (10kLs - .01) = .1009 and by equation 5-11 the operating time is then

$$t_2 = (823.9)\,[-1 \exp(.1009 \ln 10)\,]^{.903} = 245 \text{ h}$$

(6) Wasting the [(62.4) (387.5) (306)] lb of 100 °F water in the reservoir at 110 °F discharge temperature, or a 10 °F rise at q = 2 MBtuh extends the operating time (equation 5-1) by

$$t_3 = (62.4)\,(387.5)\,(306/2)\,(10/10^6) = 37 \text{ h}$$

(7) The total utilization time is then

$$t = 250 + 99 + 245 + 37 = 631 \text{ h}$$

(8) For the same reservoir but full of water F_0 = .2038 instead of .1993, S* = 400 ft^2 instead of 387.5ft^2, and from 32 °F to 50 °F the operating time is by comparison with (4) above

$$t_1 = 99(2038/1993) = 101 \text{ h}$$

Similarly, from 50 °F to 100 °F and by comparison with (5) above

$$t_2 = 245(2038/1993 = 251 \text{ h}$$

and above 100 °F by comparison with (6) above

$$t_3 = 37(400/387.5) = 38\ h$$

(9) The total utilization time is then t = 251 + 38 = 289 h if initially at 50 “F and 289 + 101 = 390 h if initially at 32 “F. This shows that the operating time of a 40/60 iced reservoir is increased by 342 h and increased by 241 h or 62 percent when compared to a water reservoir initially at a temperature of 32 “F that has no ice.

5-6. Cooling towers.

a. General

(1) The cooling tower circulating water systems for hardened facilities will generally be of a closed-loop design utilizing cooling towers, storage tanks, basins, pumps, filters, heat exchangers? and water treatment facilities. The tower construction, heat sink storage tank size, and system configuration will be determined by the facility tactical operating scenario.

(2) Figure 5-1 depicts a hardened facility cooling water system with both hardened and unhardened towers. Air for cooling towers located within areas protected by blast doors and blast valves will be separate from air used in primary protected areas. The towers will be separate from air used in primary protected areas. The towers will be surface mounted or protected below grade to escape the blast wave. The towers will have sufficient elevations above the settling basin to permit gravity flow of cooling water from the tower to the hardened basin.

(3) During normal operations, circulating water will be delivered from the cooling towers to the settling basin and pumped through the various heat exchangers in the facility.

(4) During the attack period, cooling tower fans will be shut down. Valving will be arranged to provide cooling water on a recirculation basis from the hardened heat sink storage tank with return back to the heat sink storage tank. This will provide cooling water that is independent of the cooling towers to meet attack mode heat rejection requirements.

(5) Prior to cooling tower startup following the attack period, towers and drain piping will be flushed with water from the heat sink storage tank to remove major quantities of dust and dirt to prevent plugging the system. All flushing water will flow into the dust disposal basin where it will be pumped by sand pumps to a disposal area outside the power plant. The water discharged from the disposal basin will be monitored by density meters. When the density reaches a predetermined level, the cooling towers will be returned to service.

(6) The cooling tower circulating water discharge will then be diverted through 2 micron polishing filters to remove excess dust and prevent fouling the heat exchangers. The dust removed by the filters will be backwashes to the disposal basin and will be discharged by sand pumps to the disposal area.

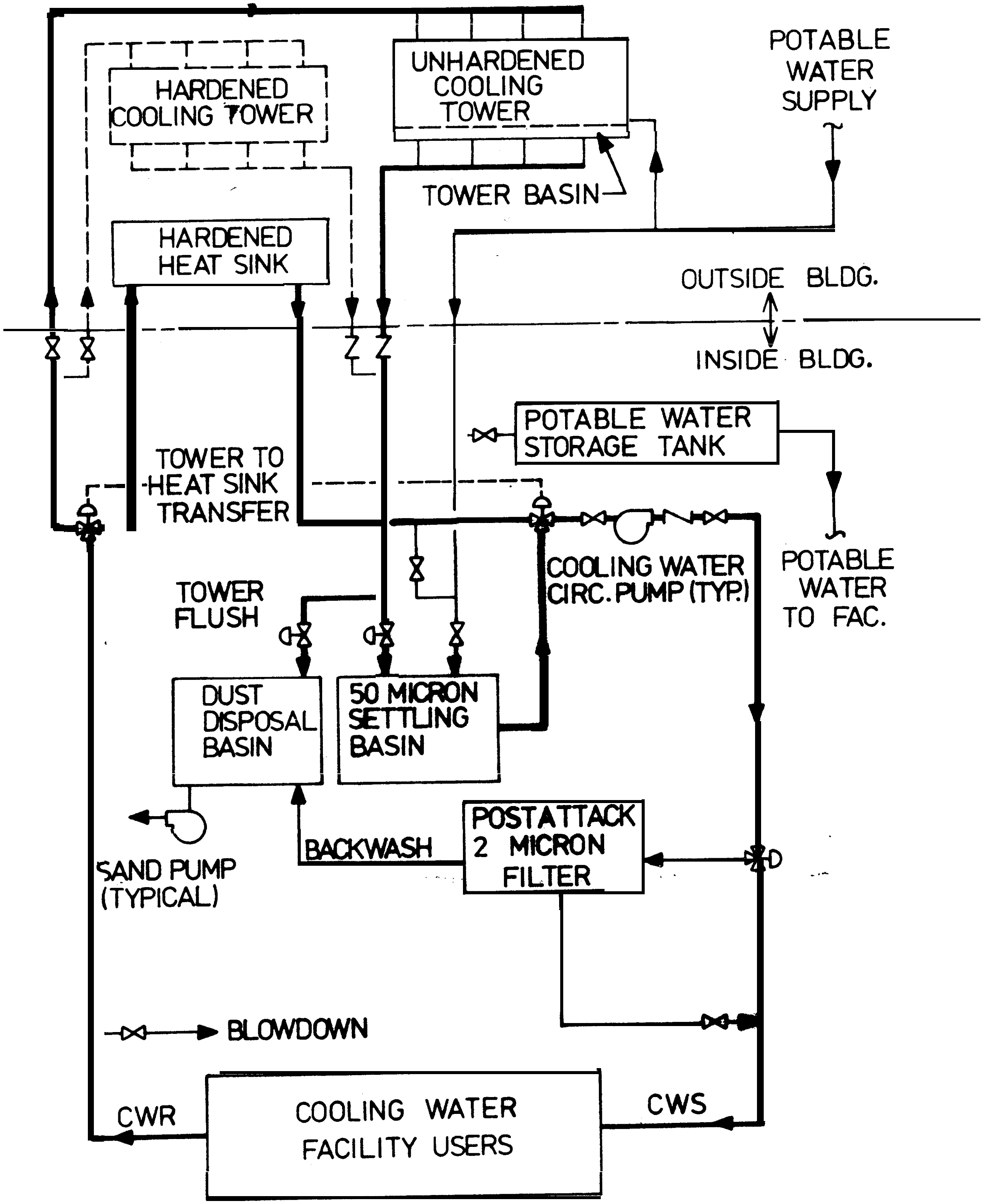

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Figure 5-1. Typical hardened cooling water system.

b. Cooling towers.

(1) Cooling towers till be of the counterflow or crossflow, spray-filled type, desired to withstand the effects of an attack. Towers will generally be cylindrical and constructed of reinforced concrete to resist severe shock loading on the aboveground tower structure, including all internals of the tower, and fan overspeeding due to weapon effects such as the blast wave. Towers exposed to direct thermal radiation will have critical parts of the system shielded from this effect.

(2) Sufficient water will be stored in heat sink storage tanks to provide for cooling tower makeup for the entire post-attack period, to provide water to flush dust from combustion air dust collectors, and to provide cooling while towers are shut down.

(3) Consideration will be given to using hardened wells, rather than hardened storage tanks, to provide for cooling tower makeup and domestic water requirements for the entire post-attack period. Wells will be used if aquifers with adequate flow capacity are available. Wells must remain a dependable source of water under all weapons effects. Where possible, two or more aquifers with flow from different directions, and separated as much as practicable will be tapped to preclude complete loss of water if one aquifer is damaged during attack. Makeup water used during normal operation will be from domestic supply or wells located on the site.

(4) Where cooling towers are required to operate in extreme winter conditions, provisions will be made to prevent freezing and ice buildup on the tower. Hardened facilities will generally have an indoor collection basin as part of the heat sink/cooling tower configuration. Additional freeze protection features, such as thermostatic cycling of the cooling tower fans controlled by leaving water temperature and reversal of the fans' direction to melt ice buildup on the tower fill, will be evaluated and incorporated into the design where necessary.

(5) Cooling towers will have fire protection system designed in accordance with NFPA 214.

c. *Basins.* Hardened concrete basins include the tower basin, settling basin, and disposal basin.

(1) The tower basin will be sized and have a flow pattern such that dirt, dust, and fallout will not settle within it during flushing of the spray towers. Flushing valves will be installed between the tower basin and the settling and disposal basins. Valves will be positive shutoff type, capable of being automatically positioned to divert water flow to the disposal basin. Valves will be constructed of materials that will not corrode or foul from the dust and dirt of a weapons blast.

(2) The settling basin will have sufficient storage capacity above the minimum suction level of the circulating pumps to provide system flow requirements for a minimum of five minutes. The basin will be of a size, design, and flow pattern to permit settlement of all dust and dirt particles having a specific gravity of 2.5 and having a size of 50 microns or larger.

(3) The disposal basin will have sufficient capacity to permit intermittent operation of the sand pumps.

d. Pumps.

(1) All pumps installed in the cooling tower circulating water system will be inherently capable of withstanding ground shock or will be dynamically mounted to reduce the shock to an acceptable level. Where possible, commercial pumps will be used if they have demonstrated resistance to the required input shock spectra. Pumps that are dynamically mounted for shock isolation will have expansion joints capable of accepting the full differential movement.

(2) Particular attention will be given to design of the pump frame, mountings, base-plates, and casings and to hold-down bolts and their proper torque specifications. Barrels of vertical pumps will be as short as possible. The use of vertical-frame pumps with excessive cantilevering in frame design will be avoided.

e. Cooling water filters. Filters used for post-attack and post-blast removal of dust and fallout from cooling tower water will be capable of removing all particulate matter larger than 2 microns and will be suitable for operation with chemically treated cooling water. Filter assemblies will be constructed of steel; cast-iron will not be used.

5-7. Radiators.

a. A finned-coil heat exchanger (radiator or air cooled condenser) exposed directly to the ambient air may be used above-ground or in a buried chamber to dissipate heat generated by prime movers, chillers, etc., in underground facilities.

(1) An underground radiator or air cooled condenser that utilizes auxiliary fans to draw (or force) cooling air from the outside through the coil and exhaust the rejected heat to the outside offers a greater degree of protection against the elements (dust or sandstorms) and weapon effects but requires more fan

horse-power for air movement. Blast closure devices, debris shields, and dirt traps in the supply and exhaust ducts will increase the level of protection.

(2) Underground radiator vaults will have provisions to wash down the radiators and pump out material deposited on the coil surfaces. The material poses no contamination problem for the cooling liquid but does reduce thermal efficiency of the radiator coils.

(3) An average air velocity of 1,500 fpm through the core, as measured by an anemometer in front of the core, is recommended. This air velocity causes a slight hum or noise, but the noise is not objectionable.

b. The fan will be operated at the speed necessary to obtain the recommended 1,500 fpm air velocity but will never be operated at more than 12,500 fpm speed when using a centrifugal type.

(1) For larger fan sizes the fan speed will fall below 1,150 rpm, the lowest recommended speed for directly connecting the fan to the motor. Fan speeds below 1,150 rpm will require provisions for reducing the electric motor shaft speed to the desired fan speed. For small installations using centrifugal type fans, the fan will be directly connected to the motor. Specifically designed propeller-type fans will be operated at higher speeds and direct-connected to electric motors running at 1,750 or 1,150 rpm.

(2) A centrifugal fan operating at a peripheral speed of 10,000 fpm is a source of noise. Sound pressure levels will not exceed 85 dBA in occupied areas. If they do, the noise will be reduced by lowering the fan speed, isolating the fan, or using inlet and outlet silencers. An average air velocity of 1,500 fpm can be obtained using relatively large fans running at lower than 10,000 peripheral fpm. A large fan running at a low speed is more practical than a small fan running at a high speed.

(3) Propeller-type fans require less power than centrifugal types. For best results they will be used as a blower-type fan and will be located from 6 inches to 10 inches in back of the core and have a shroud. Their cost, including installation, is generally greater.

(4) Forced draft produces more turbulence than induced draft, thus increasing the heat transfer rate. Induced draft provides more uniform airflow and less turbulence, handles hot air, and requires from 1 to 8 percent more fan horsepower.

(5) Values for the average power required to drive engine radiator fans are given below as a percent of the engine hp. These values are considered reasonable but may vary with coolant temperatures and ambient air temperature.

Engine HP	*Percent Engine HP*
100 or less	5.0
100 to 500	4.0
500 to 1,000	3.0
1,000 to 1,500	2.0
1,500 to 2,000	1.5
2,000 to 3,000	1.3
3,000 or more	1.0

(6) Two-speed motor drives are frequently used to save fan power when the maximum cooling effect is not required, because power requirements decrease faster than the degree of cooling. At half-speed, fans will produce 50 percent or more of total cooling capacity but will require only 20 percent of the power needed for full-speed operation. Control of airflow is another method of modulating the cooling affect. Use of two-speed motors is preferable if the heat dissipation requirements frequently vary.